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Indian Standard (Reaffirmed 2012)

GUIDE FOR SELECTION, INSTALLATION AND MAINTENANCE OF AIR COMPRESSOR PLANTS WITH OPERATING PRESSURES UP TO 10 BARS

(First Revision)

(Incorporating Amendment No. 1)

Compressors Sectional Committee, EDC 62; Accessories and Guide for Selection of Compressors, Subcommittee, EDC 62 : 3 [Ref : Doc : EDC 62 (3382)]

1. Scope — This standard covers :

a) the general method of assessment of compressed air requirement for any industrial plant, at the normal industrial operating pressures up to 10 bars;

b) general factors governing the selection of a particular type of compressor;

c) selection of the right location and method of installation;

d) accessories;

e) leakage losses and remedies against leakage;

f) maintenance of compressor plant;

g) safety devices; and

h) layout of piping from compressor to points of use.

2. Terminology — For the purpose of this standard, the definitions given in IS : 5727-1981 'Glossary of terms relating to compressors and exhausters (*first revision*)' shall apply.

3. Operating Pressure — Compressed air driven machines are generally designed for operating pressures of up to 10 bars. The normal industrial compressors operate at pressures of 6 to 8 bars. Where nothing is mentioned in this standard, the operating pressure shall be taken as 6 to 8 bars.

4. General Method of Assessment of Compressed Air Requirement

4.1 *Air Consumption of Pneumatic Tools* — Air consumption per unit in m^3/min at an operating pressure of 6.0 ± 0.5 bars for various pneumatic machine tools is shown in Table 1.

4.2 *Assessment of Maximum Air Consumption* — The purchaser shall list out the various pneumatic tools together with their numbers in each shop. Knowing the air consumption per unit, the maximum air consumption in m^3/min may be assessed for each shop. If some of the pneumatic machines are worn out due to use, provision shall be made for increased air consumption of such machines. Most of the machines operate at 6.0 ± 0.5 bars and the consumption of each class of machine is based on this pressure. For variation in pressure, the air consumption should be adjusted.

Note — The values mentioned do vary with type of machine, duty factor and design of manufacturer. There are manufacturing tolerances for air consumption of a particular machine which should also be taken into consideration.

TABLE 1 APPROXIMATE AIR CONSUMPTION OF PNEUMATIC MACHINE TOOLS

Description of Pneumatic Machine	Free Air Consumption at 6.0 ± 0.5 bars m^3/min
Core blower	0.65
Rammer, bench (2.7 to 5.5 kg)	0.30 — 0.65
Rammer, floor type (8.5 to 11.4 kg)	0.35 — 0.8
Moulding machine	0.70
Sand blast (shot) :	
8 mm jet	3.00 (at 4.2 bars)
9 mm jet	4.30 (at 4.2 bars)
11 mm jet	5.80 (at 4.2 bars)
13 mm jet	7. 0 (at 4.2 bars)
Blow gun	0.50

(*Continued*)

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TABLE 1 APPROXIMATE AIR CONSUMPTION OF PNEUMATIC MACHINE TOOLS — *Contd*

Description of Pneumatic Machine	Free Air Consumption at 6.0 ± 0.5 bars m^3/min
Fettling grip or vice	0.003 5 per operation
Hot miller (1 or 2 cutters)	0.9 per cutter
Air hoist :	
0.5 t	2.00
1 t	2.50
5 t	5.80
Chipping hammer :	
light (up to 2 kg)	0.35 — 0.50
medium (up to 4 kg)	0.50 — 0.70
heavy (up to 6 kg)	0.75 — 1.00
Wood deck caulker	0.70
Riveting hammer:	
20 — 25 mm hot rivets	0.68 — 0.73
31 mm hot rivets	0.82
34 mm hot rivets	1.00
Rivet buster, heavy single blow type	0.10 per rivet
Rivet cutter, rapid blow type for rivets up to 20 mm	1.00
Riveter, single below ('One shot' for aircraft rivets)	0.18 for 100 rivets
Riveter, staybolt for 25 mm copper staybolts	1.00
Chipper, weld flux	0.75
Sander	0.30
Tube cutter for tubes:	
62 mm	1.35
62 — 100 mm	1.65
Tube expander for tubes up to:	
62 mm	1.35
75 mm	1.65
100 mm	1.90
Scaling hammer :	
valveless, for surface work	0.20
for large boiler tubes	0.60
Grinders :	
13 — 20 mm dia wheels	0.25
up to 50 mm dia wheels	0.75
up to 100 mm dia wheels	1.20
up to 150 mm dia wheels	1.30
up to 200 mm dia wheels	1.50

(Continued)

TABLE 1 APPROXIMATE AIR CONSUMPTION OF PNEUMATIC MACHINE TOOLS — *Contd*

Description of Pneumatic Machine	Free Air Consumption at 6.0 ± 0.5 bars m^3/min
Angle grinders and polishers	0.75
Drilling machine for:	
6 mm holes in steel	0.37
9 mm holes in steel	0.45 — 0.60
13 — 20 mm holes in steel	0.75 — 0.90
22 — 25 mm holes in steel	1.00 — 1.20
32 mm holes in steel	1.30 — 1.75
38 mm holes in steel	1.50 — 1.60
50 mm holes in steel	1.65 — 1.80
75 mm holes in steel	1.80 — 2.40
For wood boring, air consumption for the immediately smaller size shall be taken. For reaming and tapping in steel, air consumption for the next larger size shall be taken.	
Wrenches (rotary type) for:	
7 mm nuts	0.15 — 0.25
9 mm nuts	0.45 — 0.50
13 — 20 mm nuts	0.75 — 1.00
22 — 25 mm nuts	1.00 — 1.20
Wrench, impact for nuts up to:	
20 mm	0.60
32 mm	1.10
Wrench, tapping	0.45
Saw, 150 mm dia	0.60
Air chuck or arbor	0.003 per operation
Air lift pump	Considerable variation according to prevailing conditions
Sump pump 400 — 1 000 l/min	1.50 — 3.60
Air press	0.007 per operation
Air motors:	
Up to 1 kW	1.2 to 1.35 per kW
1 up to 5 kW	1.2 per kW
over 5 kW	1.0 per kW
Air cylinder	0.15 per meter — tonne of lift
Drill sharpener:	
Small	1.50
Large	3.60
Forging hammer (power):	
50 kg	1.80
150 kg	3.90
250 kg	5.70

(*Continued*)

TABLE 1 APPROXIMATE AIR CONSUMPTION OF PNEUMATIC MACHINE TOOLS — *Contd*

Description of Pneumatic Machine	Free Air Consumption at 6.0 ± 0.5 bars m^3/min
500 kg	9.60
1 t	16.00
Concrete breaker:	
35 — 40 kg weight	1.65 — 2.2
25 kg weight	1.35
15 kg weight	0.90
Pile driver	1.80
Spike driver	1.80
Stone tool for:	
Lettering and light carving	0.17
Medium dressing	0.30
Roughing and bushing	0.40
Stone surfacer, for large blocks	0.90
Spike puller, per spike	0.10
Cement gun:	
Small	3.00
Medium	4.80
Grouting machine	Variable up to 3.00 at 2.5 bars
Internal vibrator, internal diameter:	
62 mm	1.10
75 mm	1.50
112 mm	2.00
140 mm	2.50
Shuttering vibrators:	
3 kg	0.18
4.5 kg	0.30
6 kg	0.45 — 0.60
Concrete compactor	0.18 — 0.30
Rock drills :	
Drifter drill (cradle mounted)	
75 mm	3.70
88 mm	4.80
100 mm	5.70
Wagon drill with:	
100 mm drifter	6.30
88 mm drifter	5.20
Hand hammer drill (Jack hammer):	
14 kg weight	1.50
17 kg weight	1.90
22 kg weight	2.25
30 kg weight	2.70
Plug drill	0.90

(*Continued*)

TABLE 1 APPROXIMATE AIR CONSUMPTION OF PNEUMATIC MACHINE TOOLS — *Contd*

Description of Pneumatic Machine	Free Air Consumption at 6.0 ± 0.5 bars m^3/min
Sinker drill:	
75 mm	3.75
88 mm	4.80
Stopping drill:	
Light	2.00
Heavy	3.75
Dustless dry drill:	
88 mm	5.40
75 mm	3.60
Underwater drill	2.50 — 2.70
Auger drill, for coal	2.24
Coal cutter, percussive:	
62 mm	3.00
88 mm	4.30
100 mm	5.40

Note 1 — The figures given are based on operation at sea level; for higher altitudes a slightly higher allowance per tool is necessary.

Note 2 — The air consumptions given above are subject to variations in different circumstances, such as the nature of the work, condition of the tools, intermittent use.

4.3 *Utilization Factor* (*Also Referred to as 'Use Factor' or 'Load Factor'*) — Ratio of total output over a period to total output when compressor operates at full load continuously over that period. In any shop where a large number of the same type of pneumatic machines are used, all the machines may not necessarily be functioning simultaneously. Hence, a use factor shall be assessed for each set of tools for correct assessment of average air consumption. This can vary between wide limits. In many cases, it is possible to fix the factors by work study. For shops using only 2 or 3 numbers of one class of machines, the use factor may be high. In cases, where use factor for each class of machines cannot be estimated, as an approximation it is common practice to use an overall factor of 50 percent of the total consumption of all machines of the plant. Even in these cases where classes of machines are one or two only, a higher use factor shall be used for such machines. Appendix A illustrates the estimation of free air requirements, taking into account the use factors.

4.3.1 *Stand-by plant* — Where compressor units are used for industrial purposes, stand-by units should receive serious consideration particularly where the compressor is vital to the production of the factory. Compressor makers aim to supply a machine, which under continuous operation will be reliable, but there shall come a time when repairs are necessary. Frequently, a small compressor as an adjunct to the main machine would serve to stand off the main machine occasionally, and if an automatic starting and stopping switch may be incorporated with the smaller machine, it could, at times of exceptional load, step in automatically and assist the main machine.

4.4 *Average Consumption of Air* — From the maximum consumption of air and the utilization factor for each class of tool, the average consumption of air for each shop may be estimated (*see* Appendix A, for examples) and hence for the whole plant. The leakage losses in pipelines and other connections to pneumatic tools should be kept to a minimum. However, a provision of 5 percent of the total average capacity requirement should be made to cover the leakage losses. Compressed air is normally used for many of the purposes not accounted for in the estimate. It is recommended that a margin shall also be provided for future plant expansion. The estimated air consumption shall be 115 percent of the calculated average air consumption. With this capacity the purchaser shall proceed with the selection of compressor.

5. General Factors Governing the Selection of Compressor

5.1 *Types of Compressors* — A brief description of the characteristics of various types of compressors commonly used is given below. A selection chart, Fig. 1, is included as a guide for selection.

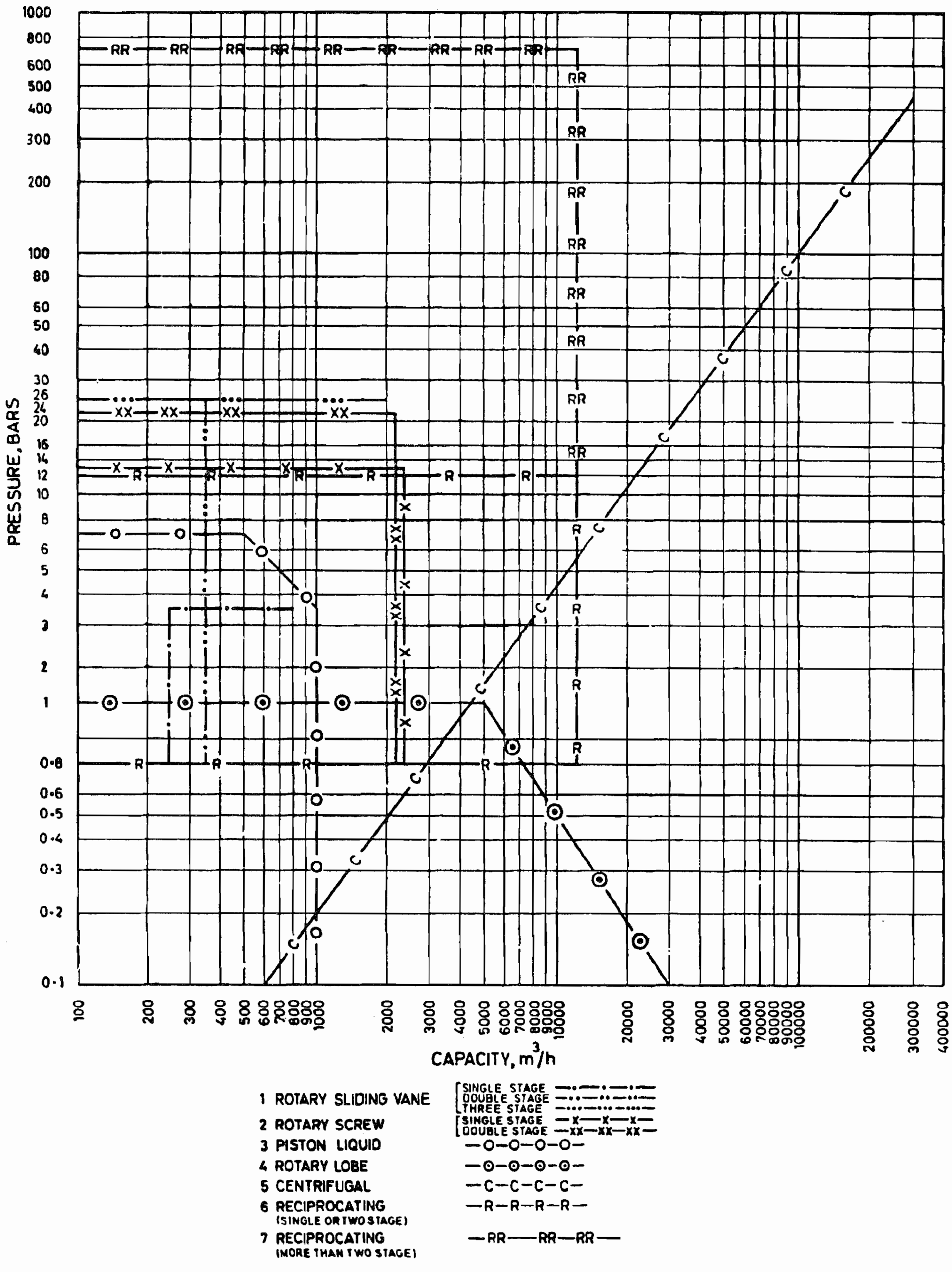

FIG. 1 SELECTION CHART FOR AIR COMPRESSOR

5.1.1 *The Reciprocating compressor* — Reciprocating air compressor is an age-old type, refined to a fine degree over the years. The various arrangements for reciprocating compressors are enumerated in IS : 5727 - 1981 'Glossary of terms relating to compressors and exhausters (*first revision*) (*see also* Fig. 2).

a) *Single cylinder*:

  i) Vertical, single or double-acting; and

  ii) Horizontal, usually double-acting.

b) *Two-cylinder*:

  i) Vertical in-line, single or double-acting;

  ii) V-type, single or double-acting;

  iii) Horizontal and vertical, usually double-acting;

  iv) Horizontally opposed, single or double-acting; and

  v) Horizontal duplex, usually double-acting (this arrangement essentially consists of two compressors side by side, with a common crankshaft).

c) *Three cylinder* — One vertical, two angled, usually at 60°, either side from the vertical, single or double-acting, usually called 'W-form'.

d) *Four-cylinder*:

  i) Semi-radial, two cylinders horizontally opposed, two at 60° upward from the horizontal;

  ii) Opposed, two pairs of horizontal cylinders on a single crankshaft;

  iii) V-type, two cylinders in each bank; and

  iv) Horizontal duplex compound. This essentially consists of four separate compressors arranged as, the four legs of an 'H' with the common crankshaft as the cross bar of the 'H'. It was an older design for very large machines and now is rarely seen.

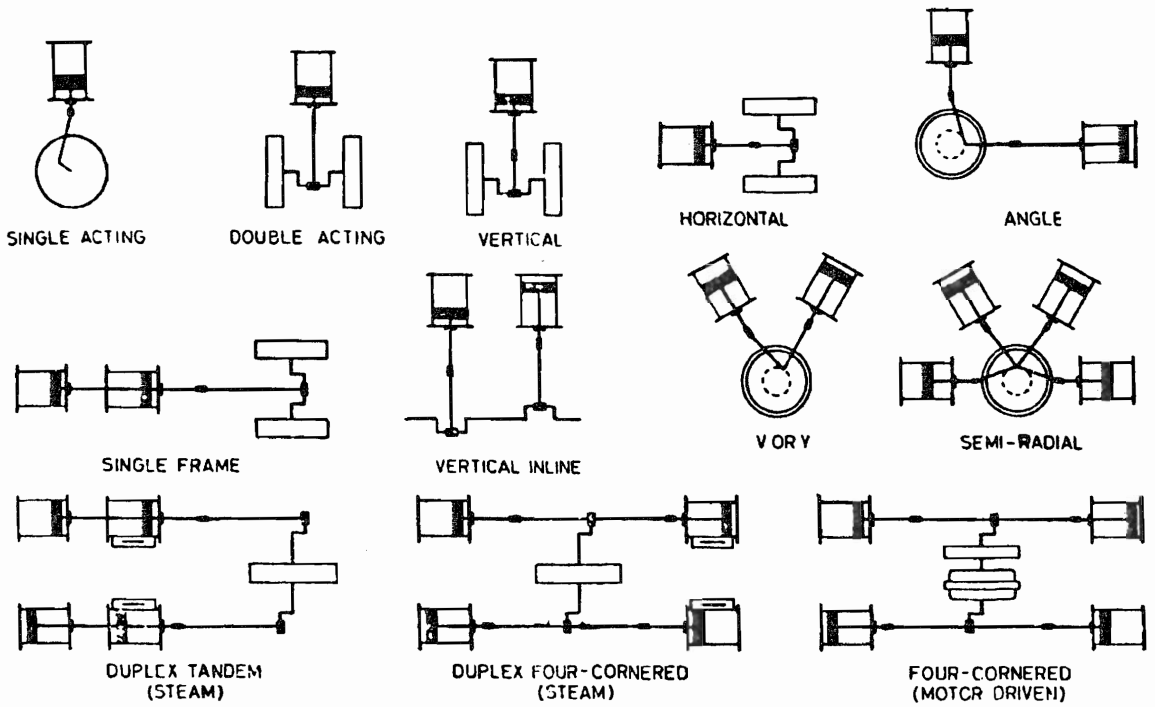

FIG. 2 COMMON ARRANGEMENTS OF CYLINDERS OF RECIPROCATING COMPRESSORS

5.1.1.1 *Efficiency and power consumption* — From the design point of view, a number of efficiencies, such as isothermal, adiabatic, volumetric and mechanical efficiencies may be considered. Similarly, in the case of power consumption, different aspects, such as isothermal and isentropic specific power consumptions may be considered. But the user is mainly interested in the overall operating efficiency which is reflected by the specific power consumption. The specific power consumption is stated in watt hours or kilowatt hours per cubic metre of free air delivered. Even though the rate per unit of electric energy varies from state to state, the specific power consumption brings out the cost factor for supply of compressed air. One can see that this is an important factor, since the cost of power which is the largest item of expense in the operation of a compressed air plant pay amount to the capital cost of the plant in a fraction of total life of the compressor.

5.1.1.2 *Single and double-acting types and piston speed* — A single acting compressor is generally characterized by the trunk type, automotive style, piston driven directly from the connecting rod and compressing on one side only. A double-acting compressor, however, has a double-acting piston driven by a piston rod, extending through a stuffing box to a crosshead which in turn, is driven through a connecting rod from the main crankshaft.

In a single-acting compressor, since, out of the two strokes in a revolution, compression takes place only during one stroke, it is used for small capacities. Also, the following characteristics are noteworthy:

a) Cylinders are commonly air-cooled;

b) The machine is compact and less costly because there is no crosshead, piston rod, stuffing box, etc;

c) Suitable for delivery as a complete package unit, including accessories;

d) Such machines are readily installed at an out-of-the-way location;

e) Very well suited for applications where air demand is infrequent or intermittent, where machine is unattended to for long periods, for construction jobs, and for temporary installations; and

f) Also suited for applications where weight and space are both critical; such as shipboard use or mining operations.

In a double-acting compressor, for the same speed and cylinder volume, air delivered is double than that of single-acting compressor, hence large-capacity machines. Other characteristics are:

a) cylinders are commonly water-cooled; and

b) normally heavy duty and continuous duty compressors are most widely used in various industrial installations.

Higher piston speeds while reducing the size of the compressors contribute to greater wear of piston rings, cylinder/liner and packing.

The piston speeds vary largely from compressor to compressor. The small capacity compressor do have piston speeds varying up to 350 m/min. Similarly, the piston speed for large capacity compressors vary up to 300 m/min.

5.1.1.3 Two-stage machines have an advantage of lower discharge temperature over single-stage types. For a single stage industrial compressor the discharge temperature is likely to be 240°C and with water-jacketing 205°C, where as for a two-stage type the discharge temperature is likely to be 140°C and even if intercooling is not efficient, maximum temperature is not likely to exceed 160°C. Lower discharge temperature are generally conducive to better compressor performance and higher efficiency. For quite small capacities, however, even up to 7 bars the use of two-stage compressor may not be fully justified since cost of two stages will be more and power saving marginal, but it would be economical for large capacities.

Both water and air-cooled types are possible up to about 100 kW. The selection shall depend on availability of cooling water and power consumption. Large compressors are invariably water-cooled.

5.1.1.4 *Oil-free compressors* — Non-lubricated compressors are available where completely oil-free air is required. These compressors are normally vertical in line or V-type compressors of the crosshead type. Lubricating oil for the compressors is avoided by the use of Teflon (tetrafluoroethylene) or other non-lubricating material for piston rings and piston rod packing rings. These compressors have slightly less volumetric efficiency than the equivalent compressors with normal cast iron piston rings. It is desirable that the cylinder valves have special Teflon inserts or stainless steel constitution.

Pistons are designed in such a way that they are guided by wear rings while reciprocating thus imposing minimum loads on the cylinders. It is also desirable to have cylinders and piston rods of good finish to reduce wear of piston rings and packings. Such compressors should have a sufficiently long distance piece to prevent oil carryover from the crank case.

5.1.1.5 *General* — Horizontal balanced, opposed compressors have better accessibility of the parts and are best suited for the large sizes, the construction by having cranks phased at 180° makes better equilization of the load and compressors perfectly balanced and free of vibration. Vertical compressors save space, facilitate balancing and have ease in installation and less wear trouble. Horizontal compressors on the other hand have better accessibility of parts and are best suited for large sizes or where head room is limited. The duplex construction by having cranks phased at 90° makes for better equalization of load than tandem type. Usually the capacity required, available space and availability will influence the choice of the particular type of compressor.

It is recommended that a non-return valve shall be fitted in the delivery line of each compressor in case they are connected to a common head to prevent back running of compressed air.

5.1.1.6 Both splash type as well as forced lubrication systems are used. It would be advisable to use forced lubrication for heavy duty industrial applications. In such cases, the manufacturer should provide suitably designed lubrication system for oil injection.

5.1.2 *Rotary compressor* — In all rotary compressors, ports in the housing replace valves and rotors replace pistons. These compressors have less number of wearing parts and require smaller foundations and have lesser vibrations. They give a continuous discharge of air and can be directly coupled to high-speed prime movers. These compressors need a lower starting torque. A non-return valve is also fitted in the delivery pipe to close off the system pressure from the rotor chamber during unloaded periods. Generally this type is used in the capacity range up to 5 000 m^3/h at an operating pressure of 20 bars. These type of compressors are available for pressure up to 35 bar.

5.1.2.1 *Dry screw compressors* — These are high-speed machines and this higher speed operation necessitates the use of suction and discharge silencers and other means to limit the noise level within a tolerable level. If the prime mover is a steam turbine the use of this type is particularly economical. The air delivery is usually completely oil-free unless the compressor is of special design when oil is injected for cooling and sealing of the clearance space (*see* Fig. 3).

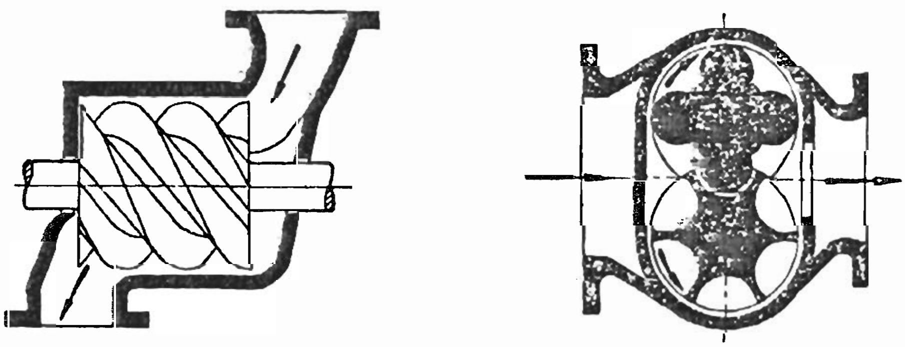

FIG. 3 ROTARY SCREW COMPRESSORS

5.1.2.2 *Oil injected rotary screw compressor* — These compressors consist of two rotors rotating in a fitting housing. One rotor is called male rotor and other is called female rotor. The rotation of two rotors brings about the same periodic increase and decrease of working space as in the movement of piston in reciprocating compressors.

Injected oil passes with the air into the receiver and is separated out through efficient air/oil separator. The oil is continuously cooled by oil cooler.

Oil injected during compression in the rotor housing performs the following function:

a) Lubrication of bearings, gears and other parts;

b) Cooling of air while it is being compressed and sealing of lobe clearances; and

c) Balancing of axial loads.

These compressors have following advantages:

a) Smaller size and lighter weight,

b) Low maintenance cost as there is no wear and tear parts in the machine,

c) Low discharge temperature of air,

d) Low starting torque which eliminates clutch in engine-driven compressors, and

e) Low foundation cost when use as stationary compressor.

5.1.2.3 *Rotary vane compressor* — The compressors consist of simple vane type, rotors enclosed in a stator body having a low pressure and a high pressure bore for two-stage machines (*see* Fig. 4).

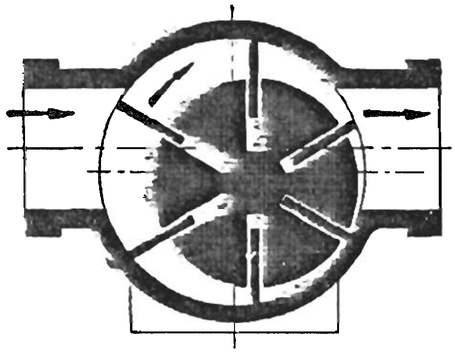

FIG. 4 ROTARY VANE COMPRESSOR

The rotors are eccentrically located in the bores of the stator body and sliding vanes, inserted in longitudinal slots in the rotors, create sealed segments of varying capacity, depending on the position of eccentricity of the segment. Inlet ports are positioned so that air is sucked into the stator bores when the gap between rotor and stator bores is increasing. When the vanes are fully extended, the segment volume is at maximum capacity. A slight further movement seals the filled segments from the intake port. Now the rotation of segment towards the discharge ports decreases its volume because of the eccentric location of the rotor in the stator body. As the segment reaches the discharge port its volume is minimum, thus the full compression is reached. Slight rotation of the segment further releases the compressed air through the discharge ports. This cycle is repeated.

There are two common types of rotary sliding vane compressor:

a) oil-flooded with non-metallic blades, and

b) water-cooled with steel blades.

In the oil-flooded rotary compressor oil is injected into the stator at both stages and performs the following functions:

a) Lubrication of bearings, gears and other parts;

b) Cooling of air while it is being compressed; and

c) Essential sealing of all clearances.

Injected oil passes with the air into the receiver and is separated out through filters. The oil is continuously cooled by an oil cooler. Although oil-flooded rotary compressor has higher power consumption due to its requirement to pump larger quantities of oil in circulation at higher pressures, the compressor has the following advantages:

a) Smaller size and lighter weight owing to its high speed,

b) Low maintenance cost,

c) Low discharge temperature of air

d) Low starting torque which eliminates clutch in engine-driven compressors, and

e) Low foundation cost when used as stationary compressor.

It should be noted that this compressor requires a low viscosity (Grade SAE 10 W as per IS : 496-1982 Specification for automobile engine lubricants). Lubricating oil and it is necessary to have very efficient oil separating system at the discharge end of the compressor.

These compressors are generally two-stage but could also be single-stage for pressures up to 7 bar and below. For higher pressures up to 21 bars compressor could be in three stages. The compressors are available in multi stages when the pressure required exceeds 21 bar.

In the water-cooled sliding vane machine:

a) lubricating oil, which is fed in droplets, serves the purpose of lubricating the bearings, vanes and other parts while also affecting better sealing; and

b) cooling of air is achieved by water-jacketing of the casing and by water-cooled intercoolers between stages.

These machines generally operate at comparatively low speeds and their advantages are:

a) reliability,

b) low maintenance cost;

c) low discharge, temperature of air;

d) low starting torque which eliminates clutch in engine-driven compressors; and

e) low foundation cost when used as stationary compressor.

5.1.3 *Roots compressors* — This is really a blower and is generally limited to a pressure of 1 bar in single stage and pressure of 2.2 bars in two-stage combination (*see* Fig. 5).

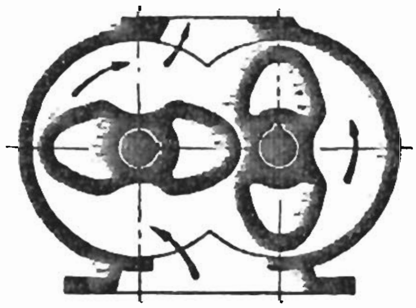

FIG. 5 ROOTS COMPRESSOR

5.1.4 *Liquid ring compressors* — The action of this compressor is similar to that of a vane compressor. Special arrangement has to be made to maintain a constant liquid level in the compressor. Further, cavitation problem limits the maximum speed at which a compressor could be driven. These are generally designed for pressures up to 5.5 bars and the consumption of power is rather high as compared to other types (*see* Fig. 6).

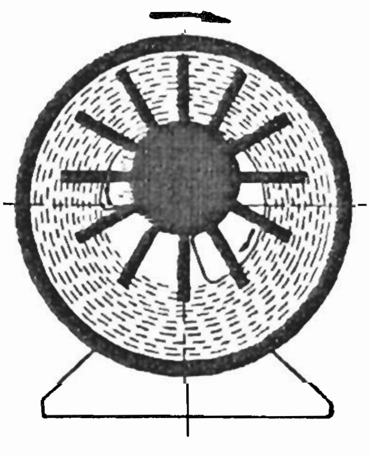

FIG. 6 LIQUID RING COMPRESSOR

5.1.5 *Dynamic compressors* — These may be single or multi-stage, radial or axial types. A turbo-compressor delivers oil-free compressed air and is economical for capacities over 500 m^3/h at an

operating pressure of 7 bars. Such compressors are used only when very large capacities are required.

6. Factors to be Considered for Selecting the Most Suitable Compressor for a Particular Requirement on the Basis of Low Fixed and Operation Costs

6.1 *Fixed Costs*:

a) *Space requirement* — Compact design of compressor and auxiliary equipment saves building space and also reduces foundation cost.

b) *Compressor accessories costs* — The compressor accessories should be available at as low a cost as possible.

c) *Foundation* — Lower weight and low unbalanced forces lead to saving in foundation cost and in lifting devices. Mounting with rubber feet on simple foundation leads to considerable saving both in maintenance and in capital cost for foundation work.

d) *Simple electrical equipment* — Compressors designed for simple standard motors and starting equipment contribute to low project cost. This also helps in getting replaceable parts for maintenance work at a low cost.

6.2 *Operation Costs*:

a) *Power cost* — A high degree of efficiency gives low energy requirements. Power cost depends on specific power consumption.

b) *Maintenance cost* — Simple easily replaceable and moderately priced wearing parts, simple design with easy access to various parts which makes maintenance work to be carried out by operating personnel feasible.

c) *Cost of supervision* — Automatic lubrications, automatic condensate water draining, reliable safety and regulating devices lead to the diminution of manual supervision and reduction in the overall cost of supervision.

d) *Lubrication cost* — Cost of lubrication in reciprocating compressors is much lower than in rotary compressors. Specific oil consumption should be low for low operating costs.

7. Location of Compressor Plants

7.1 The following factors should be considered while selecting the location of the compressor:

a) The compressor should be located near the shop which consumes the maximum compressed air, so as to keep the pressure drop and the leakage losses in the pipelines to a minimum. Further, piping costs can also be reduced considerably.

b) The location should be as far as possible be such that intake air is clean, free from dust, soot and other impurities. Where it is not possible the suction filter can be located in clean air and ducting connected to the compressor. Special suction filters are available to deal with unclean air if it becomes necessary to install the compressors in such surroundings. The change in ambient temperature does not effect FAD of compressor. However, lower ambient temperature increased the mass flowrate.

c) Where possible the soil conditions should be such as to avoid the necessity of piling.

d) Vibrations from the compressor can propagate through the walls and affect the operation of other sensitive machinery in the plant. The location should be such as to avoid the effects of vibration and if this is not possible then methods for reducing the vibration effect may have to be considered.

e) The location should be such that maximum economy in connecting power lines, cooling water and waste water piping is achieved.

f) The compressor room should be so designed as to allow space for future expansion and to provide sufficient ventilation. The ventilation arrangement should take into consideration the dissipation of the heat released by the compressor when in operation and the heat released by air cooled after the coolers. In the case of large compressors special attention may have to be given for fitting proper lifting gear.

8. Erection of Compressors

8.1 On the basis of users report on load bearing and soil conditions, the manufacturers supply detailed foundation drawings which shall be closely followed. Alternatively, the foundation may be designed by purchaser on the basis of load data given by the manufacturer. These loads shall include, apart from dead loads, dynamic and out of balance loads and their lines of action.

8.2 In the case of reciprocating compressors, complete balancing is difficult except with opposed piston types. The frequency of vibration of a compressor is normally twice its rotational speed. The natural frequency of the compressor foundation mass shall not synchronise with the natural frequency of the compressor to avoid resonance. These facts should be taken into consideration in the preparation of foundation drawings. Where necessary, the compressor is equipped with vibration damping rubber pads which may be bolted to the floor or support frame. External connecting pipes from compressor are made of flexible pipes.

8.3 The foundation of the compressor of large size should preferably be isolated from the rest of the building.

8.4 The compressor should be installed so that there are proper vertical and horizontal clearances necessary for dismantling internal parts like piston and piston rod. Also, ample space all round shall be left to permit cleaning, inspection and repairs.

8.5 Before erection, the top of foundation shall be roughened. There shall be no trace of grease or oil over this. The compressor should be erected scrupulously following the manufacturer's recommendations on levelling, alignment, grouting, etc. It shall be perfectly levelled in both the directions. While connecting piping, due care shall be taken to see that there are no strains imposed on the machine by poorly fitted piping. This can be ensured by properly aligned pipe work and by supporting pipes in the vicinity of the machine. If suction duct is long, it may be connected to the compressor through the flexible coupling.

9. Drives

9.1 Compressors may be driven by one of the following methods;

a) Direct drive:

i) Integral shaft for prime mover and compressor, and

ii) Flexible or rigid coupling between the prime mover and the compressor;

b) V-belt drive; and

c) Clutch.

9.2 *Direct Drive* — With such an arrangement there is no transmission loss. The motor speed will necessarily have to be same as that of compressor. The compressor speed is fixed at one motor speed and as such the capacity of the compressor cannot be varied to suit the requirements of the purchaser. The motor alignment should be perfectly correct, more particularly for rigid couplings to avoid undue stresses being set up. With this type the compressor and motor are usually delivered as one unit up to 100 kW, already aligned by the manufacturer by means of guide pins or a guide flange. Direct driven compressor usually shows a saving in space and the foundation is also simple.

9.3 *Flexible Coupling Between Motor and Compressor* — Here the risk of undue stresses being set up in the compressor crank shaft and effect on the air gap of the motor, due to incorrect alignment is reduced. However, alignment should be proper to increase the life of the coupling.

9.4 *V-Belt*

9.4.1 This is a common method of drive. V-belts for industrial purposes are covered by IS : 2494-1974 'Specification for V-belts for industrial purposes' for belt ratings up to 41.7 kW.

The pitch length of belts corresponding to given pulley pitch diameters and centre distances may be obtained by the formula given as follows:

$$L = 2C + 1.57\,(D + d) + \frac{(D - d)^2}{4C} \text{ and}$$

centre distance, $C = A + \sqrt{(A^2 - B)}$

where

L = pitch length of belt,

C = centre distance of the drive,

D = pitch diameter of the large pulley,

d = pitch diameter of the smaller pulley,

$A = \frac{L}{4} - \sqrt{\frac{(D + d)}{8}}$, and

$B = \frac{(D - d)^2}{8}$.

9.4.2 The disadvantages of belt drives are:

a) the tendency of many mechanics to over-tighten the belts with possible damage to bearings or even broken motor or compressor shafts;

b) the necessity of replacing worn belts, but a well-designed modern V-belt drive has a long life; and

c) a belt power loss of about 5 percent.

9.4.3 The advantages with the use of belt drives are:

a) high speed motor, generally 1 000 or 1 500 synchronous rev/min can be used, which is cheaper and more easily available and replaceable in the event of an accident or failure; and

b) the speed of the compressor could be varied to a close limit suit the user's requirements. The manufacturer would be in a position to supply either a maximum-rated machines for lowest cost, or a conservatively-rated machines for longest life, and a possible higher speed and output, if increases should occur later in air requirements.

9.4.4 For long life of belts, correct alignment is essential. Belt tension should be checked and adjusted where necessary, after a few hours of operation at full speed. Belt tension shall be adjusted only when the motor is stationary and not running. Belt tension is to be checked by suspending a weight of 2 kg approx, on one of the belts mid-way between the flywheel and pulley. For correct tensioning, the belt should sag approximately by an amount equal to the thickness of the belt. Over-tensioning the belt accelerates loss of elasticity, shortens the life of the belt, and subjects the shafts and bearing to excessive loads. The cross-sections of endless V-belts are classified as A, B, C, D and E. Since more than one belt is to be used for transmission of power, matched belts should be used in order to avoid uneven distribution of load. Even if one belt is worn out in a set, the complete set shall be replaced by matched belts. Oil and grease should be prevented from coming in contact with belts. Proper guards should be fitted over the belt drive portion to prevent whipping of broken belt and causing injury to the operating personnel.

9.5 *Drive Motor* — The selection of the driving motor will be governed by the following factors:

a) *Power supply and voltage* — The motor selected should be suitable for the available power supply. Voltage and frequency fluctuations should be specified. For large motors it would be necessary to use higher voltage supply, namely, 3.3, 6.6 or 11 kV.

b) *Type of motor* — For ac supply system, the choice falls between induction motors and synchronous motors. The former are of two types squirrel cage and wound rotor. The squirrel cage construction is mechanically strong and easy to maintain as no brush gear, starting resistance, etc, are involved. The wound rotor has the advantage that starting torque can be increased, during starting by external resistance which is cut out after the motor has started.

Synchronous motors may be preferred in large ratings and have the advantage of constant speed and power factor improvement. However, for excitation and starting of these motors additional equipment like excitors, control panels and interlocks are required involving extracost and maintenance.

c) *Enclosures* — Electrical motors are constructed with a variety of enclosures, for example, screen protected, drip proof, totally enclosed fan-cooled, pipe-vantilated, duct-vantilated, water-cooled, increased safety or flame-proof. IS : 4691-1968 'Degree of protection provided by enclosures for rotating electrical machinery' specifies the degree of protection provided by enclosures for rotating electric machinery. The enclosure for the motor should be carefully selected taking into account the prevailing conditions where it is required to be installed.

d) *Ambient temperature, mean sea level and humidity* — Ambient temperature and mean sea level have a large effect on motor rating. Where humidity is high, anti-condensation devices, space heaters or drain plugs may be provided for the motors — specially in higher ratings above 50 kW. Where motors are located outdoors, it should be so specifically mentioned.

e) *Performance* — The starting torque and moment of inertia of the load to be accelerated are of major consideration. It is preferable to provide a complete starting torque-speed characteristic of the load, wherever available to the manufacturer. IS : 325-1978 'Specification for three phase induction motors (*fourth revision*)', specifies the performance requirements of induction motors.

f) *Motor duty* — The motor may have to be used either continuously or intermittently. Intervals between starts and stops have a great effect on motor ratings.

g) *Motor ratings* — The motor nameplate rating shall be such that it is capable of running the compressor at least 110 percent of rated pressure.

10. Capacity Control

10.1 With compressors driven by diesel engine, turbine or dc motors, the air output may be varied by varying the speed of the driving equipment.

10.2 Automatic start-stop operation of motor is also possible but its use shall depend on the size of the motor, the nature of the application and local conditions of power supply. As a general rule, when start-stop operations exceed 6 per hour, this method of capacity control is not used.

10.3 Unloading the compressor with the machine running is commonly adopted for capacity control of machines above 25 kW and universal for machines above 120 kW.

10.4 In the case of reciprocating air compressors driven by constant speed movers, unloading with the machine running is commonly adopted for capacity control of machines to adjust supply to varying demand. Two devices to this automatically are available on different models of machines, namely, (a) throttling the suction, and (b) keeping the suction valves of the cylinder in open position in delivery stroke by suction valve lifters. Both these power. Suction valve unloaders are preferable when the compressor has to run long hours on light loads.

10.5 A combination of automatic stop-start and load-unload control of compressor with manual selection may also be used for capacity control.

10.6 An automatic dual control, with the compressor selecting for itself whether it will run on start-stop control or at constant speed with unloading controlled by a timing device, may also be used.

10.7 *Sequence Loading and Unloading* — It is adopted for a battery of compressors. The loading and unloading range is controlled by solenoid valves and pressure switches which are pre-set to ensure that each compressor loads and unloads in the predetermined sequence. This is necessary when each compressor in the battery is not having a separate air receiver.

10.8 Reciprocating compressors above 100 kW are sometimes provided with clearance pockets which, when manually or automatically opened, reduce the capacity of the machine in predetermined steps.

10.9 Another method of capacity control is to release excess air to atmosphere. This, however, wastes power.

10.10 For rotary compressors driven by constant speed equipment, the usual method of capacity control is suction throttle.

11. Safety Devices

11.1 Low-oil pressure protection with suitable alarm is desirable and automatic trip should be fitted as an insurance against careless maintenance, wrong connection of motor with consequent reverse drive of compressor or mechanical failure of oil system.

11.2 With water-cooled compressors and intercoolers, protection against high cooling water outlet temperature is recommended. With high water temperature, efficiency of the compressor drops and may cause overheating resulting in the motor being cut out by the overload relays. Relays operating at pre-set pressure or flow of cooling water are also generally used as protection against cooling water failure.

11.3 Safety valves should be provided against abnormal rise of air pressure. When a number of compressor are installed and stop valve fitted between compressor and air receiver, safety valves shall be fitted on the compressor side of stop valve. The safety valve should be large enough to code up with the volume of air delivered by the compressor. It should be so constructed as to allow the air to escape without increasing the pressure beyond 10 percent of working pressure when the compressor is delivering the full output.

When installing the compressor it is important to see that the manufacturer has fitted the safety valve in the intercooler between the various stages. If the high pressure valves are leaking, excessive load may be put on the low-pressure cylinder and may damage intercooler in the absence of a valve.

11.4 Protection against high discharge air temperature is available by means of a pre-set figure on a thermometer or a fusible plug.

11.5 In the cases of compressor where separate starting oil pumps are provided for the lubrication of the bearings and cylinders, the oil pump motor should be directly connected with the main motor of the compressors, so that the main motor may not be started unless the oil pump is started.

12. Effect of Altitude

12.1 The increase in altitude above mean sea level, decreases the atmospheric pressure. This contributes to higher overall compression ratio when compressor is operated at an altitude as compared to sea level operation at the same discharge pressure.

The compressor capacity is governed by the clearance volume and compression ratio in the first stage compression chamber of the compressor. The capacity of the compressor decreases with the increase in the compression ratio due to higher expansion in the clearance volume of the compression chamber or in other words decrease in volume intake or decrease in volumetric efficiency.

12.2 *Rotary Screw and Vane Type Air Compressors* — For rotary air compressors, the increase in altitude has marginal effect on the capacity as the clearance volume is negligible in rotary design.

The percentage reduction in capacity for rotary air compressor, at 7 bars (gauge) discharge pressure for each 1 000 metres increase in altitude above sea level:

a) Rotary single stage	0.8 - 1.0 percent
b) Rotary two-stage	0.4 - 0.5 percent

12.3 *Reciprocating Air Compressors* — Increase in altitude has a considerable effect on the capacity of single stage reciprocating air compressors. This may be estimated by choosing the appropriate factor corresponding to the altitude and the discharge pressure from Table 2 and multiplying the sea level capacity of the unit by that factor.

12.3.1 For two stage reciprocating air compressors the effect of altitude on the capacity is marginal.

The percentage capacity reduction for two-stage reciprocating air compressors at 7 bars (gauge) discharge pressure for each of 600 metres increase in altitude above mean sea level is around 0.5 - 0.6 percent.

Note — For estimating capacity at altitudes for various type of air compressors, **12.2** and **12.3** are given as guidelines. However, for exact duration in capacity at altitudes the manufacturer may be consulted.

12.4 *Pneumatic Tools* — The air consumption of pneumatic tools at a fixed operating pressure increases with the increase in altitude. Table 3 gives the multiplying factor corresponding to **6.2** bars (gauge) operating pressure at various altitudes. The air consumption of pneumatic tools can be determined by multiplying air consumption at sea level by the factor corresponding to the altitude.

TABLE 2 EFFECT OF ALTITUDE ON THE CAPACITY (FAD) OF RECIPROCATING SINGLE STAGE AIR COMPRESSORS (BASIS OF SEVEN PERCENT CYLINDER CLEARANCE)

(*Clause* **12.3**)

Discharge Pressure Bars (Gauge)		**1.75**		**2.8**		**4.2**		**5.6**		**6.3**		**7**	
Altitude in metres	Atmos-pheric Pressure Bars (abs)	Comp-ressure Ratio	Factor	Comp-ressure Ratio	Factor	Comp-ressure Ratio	Factor	Comp-ressure Ratio	Factor	Comp-ressure Ratio	Factor	Comp-ressure Ratio	Factor
Sea Lavel	1.013	2.727	1.000	3.764	1.000	5.146	1.000	6.528	1.000	7.219	1.000	7.910	1.000
300	0.978	2.789	0.996	3.863	0.994	5.294	0.992	6.726	0.990	7.442	0.988	8.157	0.987
600	0.943	2.856	0.992	3.969	0.989	5.454	0.984	6.938	0.979	7.681	0.976	8.423	0.974
900	0.910	2.923	0.988	4.077	0.983	5.615	0.975	7.154	0.968	7.923	0.964	8.692	0.961
1 200	0.877	2.995	0.984	4.193	0.976	5.789	0.966	7.385	0.956	8.184	0.952	8.982	0.947
1 500	0.846	3.068	0.980	4.310	0.970	5.964	0.957	7.619	0.945	8.447	0.939	9.274	0.933
1 800	0.815	3.147	0.976	4.436	0.963	6.153	0.948	7.871	0.933	8.730	0.925	9.589	0.918
2 100	0.785	3.229	0.971	4.567	0.956	6.350	0.938	8.134	0.920	9.025	0.911	9.917	0.903
2 400	0.756	3.315	0.966	4.704	0.949	6.555	0.928	8.407	0.907	9.333	0.897	10.259	0.887
2 700	0.728	3.404	0.961	4.846	0.942	6.769	0.918	8.692	0.894	9.654	0.883	10.615	0.872
3 000	0.701	3.496	0.956	4.994	0.934	6.991	0.907	8.988	0.881	9.987	0.868	10.986	0.856
3 300	0.674	3.596	0.951	5.154	0.926	7.231	0.896	9.309	0.867	10.347	0.853	11.386	0.839
3 600	0.649	3.696	0.946	5.314	0.918	7.471	0.885	9.629	0.853	10.707	0.837	11.786	0.822
4 200	0.599	3.921	0.934	5.674	0.901	8.012	0.861	10.349	0.822	11.518	0.804	12.686	0.785
4 500	0.578	4.028	0.928	5.844	0.893	8.266	0.849	10.688	0.808	11.900	0.788	13.111	0.769
5 000	0.540	4.241	0.917	6.185	0.877	8.778	0.828	11.370	0.781	12.667	0.759	13.963	0.736

Note — To find the capacity of a compressor when it is used at an altitude, multiply the sea level capacity of the unit by the factor corresponding to the altitude and the discharge pressure. The result will be the actual capacity of the unit at the altitude.

TABLE 3 MULTIPLIER TO DETERMINE AIR CONSUMPTION OF PNEUMATIC TOOLS AT ALTITUDES
ASSUMED OPERATING PRESSURE OF 6.2 BARS (GAUGE)
(*Clause* **12.4**)

Altitude in Metres	Air Consumption Multiplying Factor
Sea level	1.000
300	1.031
600	1.064
900	1.097
1 200	1.133
1 500	1.170
1 800	1.209
2 100	1.250
2 400	1.292
2 700	1.336
3 000	1.383
3 300	1.432
3 600	1.482
4 200	1.594
4 500	1.647
5 000	1.753

To determine air consumption of tool at an altitude, multiply the sea level consumption by the corresponding factor.

13. Accessory Equipment

13.1 *Intake Filters* — For reliability and durability of a compressor it is absolutely essential to provide a suitable and efficient air intake filter. Excessive wear of moving parts and mal-functioning of valves is mainly caused by the abrasive effect of impurities in the air. The air intake filter for a compressor should satisfy the following requirements:

a) High separating capacity;

b) Ability to collect large quantities of impurities with minimum frequency of cleaning;

c) Low air resistance, that is, low pressure drop across filter; and

d) Robust design, as the filters are likely to be subjected to an air pulsation in the intake pipe.

Filters should be placed as close as possible to the compressor. If a silencer is fitted then the filter should be place upstream of the same. The position of the filters should be such as to facilitate easy inspection and cleaning. It is desirable if an indicating device is provided to show when the intake filter requires a change.

13.1.1 *Paper filters* — The filter element in this type consists of corrugated impregnated paper. The paper filters shall not be exposed to temperatures above 80°C or too strong air pulsations. The filtering efficiency of a paper filter is about 90 percent. With new elements, the pressure drop is about 25 to 35 mm H_2O and the pressure drop increases with usage. Generally, these are used in combination with venturi silencer to give a good damping effect. In places where the humidity is likely to be high, paper filters may not fulfil their function properly as they are likely to absorb moisture from the air.

13.1.2 *Oil wetted filter* — The filter element is made of corrugated expanded sheeting or corrugated wire mesh pressed together and the filter is kept moist with oil. The efficiency of this type of filter decreases rapidly as its oil-wetted surface dries up or catches dust. The time intervals between cleaning operations depend on the conditions of the intake air and generally cleaning is required every 50 to 100 hours. This type of filter is cleaned by blowing it through the reverse direction with steam or washing it with trichloroethylene. After cleaning proper drying and blowing compressed air through it, is necessary.

13.1.3 *Cloth filters* — Elements of these filters usually consist of woollen or nylon cloth attached to wire netting and heavily pleated in order to obtain a large filtering area. These are usually fitted on the inlet flange of the compressor. Woollen cloth filters have a high degree of filtering properties and are used with compressors meant to deliver oil-free air. These should be inspected after every 50 to 100 hours and cleaned when required by subjecting it to a reverse flow of compressed air. This type is also referred to as dry filter. When in use this type has a tendency to load up fast with the contaminants in the air.

13.1.4 *Oil-bath filters* — The filter body is arranged so that the air flow is restricted causing increase in air velocity over the oil-bath. A large proportion of impurities adheres to the oil surface and falls to the bottom in the form of sludge. As the air impinges on the oil, a mist of oil rises up the wire mesh. The fine particles of dust which are still in the air after passing over the oil-bath are collected by the oily surface of the wire mesh thereby delivering clean air to the compressor. This type may cause a pressure drop of 150 mm of water column. Filters of this type have a high accumulating capacity and can collect a large quantity of impurities generally equal to the weight of oil used. The oil used in the oil-bath should be same as that recommended for compressor lubrication.

Where the working conditions are such that the intake air is likely to be highly polluted with dust, use of oil-bath filter becomes a necessity. Further, where there is a doubt that the compressor plant is not likely to be regularly maintained it is better to specify this type of intake filter, as this will maintain good performance over a long period with very little maintenance.

13.1.5 *Combined dry and wet type filters* — In this type of filter the air is filtered for dust particles, initially by causing vortex thus separating heavy particles of dust by centrifugal action. Further, this air is passed through a regular oil-bath filter, as mentioned above. This combined filter is very effective in more dusty conditions.

13.2 *Pulsation Dampers* — With reciprocating compressors, there are two categories of pulsating flow as follows:

a) Interrupting pulsating flow from a single acting, single cylinder compressor; and

b) Power pulsating flow from a double acting, two-cylinder compressor.

In some compressor plant layouts the lengths and bends of the pipelines may be such as to develop resonance conditions. Under such conditions excessive noise and vibration will be set up and in some cases may even result in the bursting of the discharge pipeline due to the pulsations in the line having a higher amplitude. The compressor has to work against the higher pressures and therefore the power consumption is slightly more. To prevent this, a pulsation damper, which is in fact a small receiver, is mounted directly on to the discharge flange of the compressor and acts as a surge tank to minimise the pulsation in the delivery side of the compressor. Compressor manufacturers will be in a position to suggest whether a pulsation damper volume bolts is necessary for a given plant, provided the user supplies them with the plant layout with all the pipeline arrangement.

13.3 *Aftercoolers* — Aftercooler is a simple type of heat exchanger and should conform to IS : 4503-1967 'Specification for shell and tube type heat exchangers'. In general, the cooled air should be brought to within 10°C of the cooling water supply temperature. In the design of aftercoolers provision must be made for the removal of water vapour condensation. The amount of water vapour to be dealt with will obviously depend on the relative humidity of the environment in which the compressor is to be operated. The *p*H value of water shall be between 5 and 6 and shall conform to IS : 8188-1976 'Code of practice for treatment of water for industrial cooling systems'.

13.3.1 Where air is required with the minimum possible moisture content dryers may have to be used on the outlet side of the aftercooler. Use of fan-cooled, radiator type, aftercoolers may be considered where the cost of water is high as these units are equally efficient in operation. Aircooled aftercoolers find application where temperature is 12 to 15°C above ambient.

In multiple unit compressor plants it is desirable to have separate aftercoolers for each compressor unit, as this will give flexibility of operation especially is one of the units goes out of order or for routine maintenance of units.

13.4 *Receivers* — It is essential to provide one or more air receivers after a reciprocating compressor, or a set of such compressors. It is normal to provide the receivers for other types also. The air receivers shall conform to IS : 7938-1976 'Air receivers for compressed air installation'. The air receiver serves the following purposes:

a) To damp out pulsations of the air delivered by the compressor (applies to reciprocating compressors only);

b) To match the flow of air with the load cycle so that varying demands can be met without the compressor regulator working incessantly;

c) To condense and trap as much as possible of the moisture and oil in the receiver; and

d) To store compressed air the capacity of the air reservior to and pulsations in a reciprocating type should be about one-tenth of the free air delivery per minute increasing to about one-sixth for smaller compressor. This should be regarded as a minimum and additional capacity should be provided depending upon the requirement to meet piece load requirements with the installed capacity of compressor installation.

Note — This applies to stationary installation only.

The receiver volume also caters to requirement of air during changeover from one compressor to the other. Thus, in case of a system provided with a start and stop control, larger volume of the air receiver is necessary.

A compressed air receiver is a vessel to store air at higher pressures than atmospheric. Thus the design, construction and inspection of compressed air receivers and pipes are governed by national statutory rules. The construction testing and inspection of air receivers should generally conform to IS : 2825-1969 'Code for unfired pressure vessels'.

In the case of special applications, such as air-soot blowing, where a continuously pumping small compressor is required to provide large quantities of air for a few minutes in each hour, it might be more economical to use standard multiple receivers rather than a specially designed large single unit.

Air receiver should be provided with a pressure gauge safety valve and a moisture drain valve.

14. Compressed Air Pipe System

14.1 To obtain maximum economy, reliability and high efficiency of compressor air pipe system, the following points should be kept in view:

a) Low pressure drop between the compressor plant and the points of air consumption;

b) Minimum leakage;

c) High degree of condensate separation throughout the system;

d) Minimum number of joints, bends and fittings in pipeline; and

e) Expansion joints are recommended on hot piping after the compressor, in case aftercoolers are not being used. However, in case of compressor for stationary applications, the compressor shall be provided with aftercooler.

The pressure drop in the pipeline with the necessary control valves up to the point of consumption should be minimum so that the pneumatic machine at the operating point is able to work at its rated pressure. If a pneumatic machine is operated, say at 5 bars with its normal working pressure at 7 bars, then the power developed by this machine is only 55 to 60 percent of the rated power.

The dimensions of the pipe system should be such that a future increase in air consumption within reasonable limits should cause no significant pressure drop which may necessitate re-laying the existing pipelines.

A general guide for selection of pipe diameter for compressed air pipelines is that the total pressure drop in the pipelines does not exceed 0.3 bars between the compressed air plant and the most distant point of air consumption. In the case of installations where operating points are distributed over a large area, a higher pressure drop up to 0.5 bars may be accepted. In addition to this, pressure drop in hose couplings and other compressed air fittings should be kept to a minimum. Often increasing the diameter of the hose couplings may reduce the pressure drop in these fittings considerably.

A prime consideration in the design of pipe system should be to prevent leakage, as this has considerable effect on the operating economy of a compressed air installation. The pipe system should preferably be designed as a closed-loop system so as to supply air from two directions around the area where air consumption takes place and from this main line branch lines should be run to different points of air consumption, to obtain a uniform supply of compressed air at all points. If heavy air consumption points are located at considerable distance from compressor, then a separate main line should be run to these points. While laying the pipelines, water separators should be provided at suitable points to drain condensate from the line from time to time. The line

should be made as straight as possible with very few bends and also with a slight slope (usually 1/40 approx) for easy flow of condensate towards the water separator. Care should be taken to see that suitable vibration damping material is used with the clamps in case of pipelines supported by clamps to ceilings so that vibrations in the pipeline carrying the compressed air are not transmitted to the building or structure to which the brackets are attached.

The following table gives the inner diameter, in millimetres, of the pipeline for a pipe length of 100 m and pressure drop of 0.1 bar for various values of free air flow and delivery pressure. To ensure a pressure drop not exceeding 0.1 bar, standard pipe sizes conforming to IS : 3601-1966 'Specification for steel tubes for mechanical and general engineering purposes' which are nearest to the tabulated values may be used. Corrections should be applied for fittings in the piping as follows:

Gauge Pressure Bar ↓ / Flow of Free Air m^3/min →	1	2	3	4	6	8	12	16	25	30	40	50	60	90
3	29	38	44	49	56	63	74	81	96	103	115	125	133	154
7	25	33	38	47	50	55	63	72	84	90	100	108	116	135
14	22	29	34	37	44	48	56	63	74	79	80	90	102	120
21	21	27	31	35	40	45	52	58	68	73	82	88	95	110

The pressure drop in the pipeline can be calculated by the following formula:

$$p = \frac{7.57 \times Q\ 1.85 \times L}{d^5 \times P} \times 10^{-10}$$

where

p = pressure drop in the pipeline, bar;

Q = air flow quantity, m^3/min (FAD);

L = length of the pipeline, m;

d = inside diameter of the pipeline, in mm, conforming to IS : 1239; and

P = initial absolute air pressure, bar.

The length L in metres used in this formula should include the equivalent pipe lengths, in metres, for pressure loss in valves, bends elbows, hose connections, etc. The equivalent lengths, in metres, for commonly used valves and fittings are shown in Table 4.

TABLE 4 EQUIVALENT PIPE LENGTHS FOR VALVES AND FITTINGS

Valves and Fittings ↓ / Nominal Inner Diameter of Pipe mm→	25	40	50	80	100	125	150
Seat valve	3-6	5-10	7-15	10-25	15-30	20-50	25-60
Angle valve	5-6	6-8	8-10	12-15	18-20	20-25	25-30
Swing check valve	2-3	3-4	4-5	5-6	7-8	10-12	12-14
Diaphragm valve	1.2	2.0	3.0	4.5	6	8	10
Gate valve	0.3	0.5	0.7	1.0	1.5	2.0	2.5
Elbow	1.5	2.5	3.5	5	7	10	15
Bend; radius = inner dia	0.3	0.5	0.6	1.0	1.5	2.0	2.5
Bend; radius = 2 × inner dia	0.15	0.25	0.3	0.5	0.8	1.0	1.5
Hose-connection T-piece	2	3	4	7	10	15	20
Sudden contraction d/D = 0.5	0.3	0.4	55.0	0.85	1.10	1.50	1.70
Sudden contraction d/D = 0.75	0.17	0.25	0.37	0.50	0.70	0.90	0.10
Reducer	0.5	0.7	1.0	2.0	2.5	3.5	4.0

As mentioned earlier, the total pressure drop, that is, inclusive of the pressure drops in valves and other fittings, should not exceed 0.3 bar. Depending on the length of the pipeline the air flow speed should be kept between 6 and 10 m/s to give a moderate pressure drop. Pressure drop can also be readily obtained from graph (*see* Fig. 7).

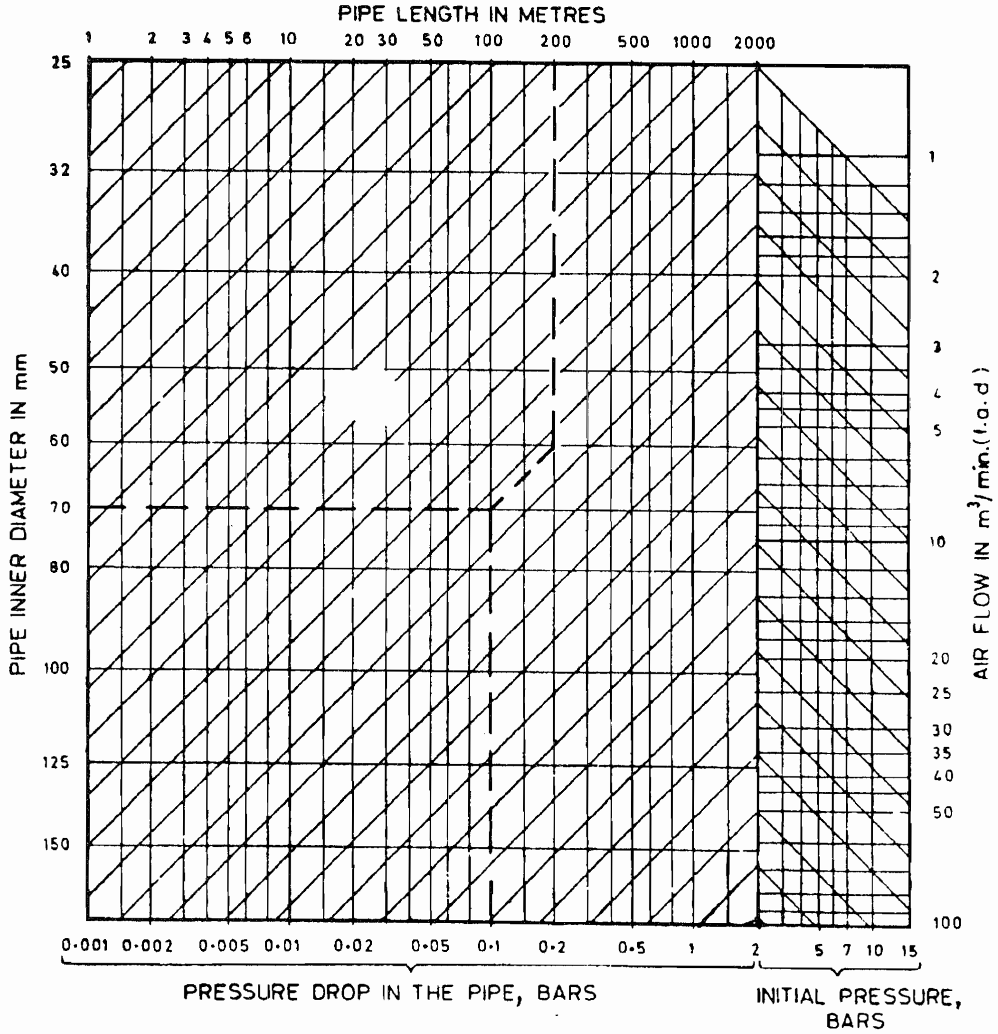

FIG. 7 GRAPH FOR CALCULATION OF PIPELINE DIMENSIONS

The initial pressure of air in bars and the flow in m^3/min are given on the right-hand side of the figure.

To use this figure select the operating initial pressure and proceed vertically to meet the horizontal line corresponding to the air flow in m^3/min. From this junction proceed diagonally upward to the boundary and from there proceed horizontally to meet the vertical line corresponding to the length of the pipe in metres (inclusive of equivalent length). From this junction proceed diagonally downwards till the horizontal line corresponding to the inner diameter of the pipe is intersected. Finally, from this intersection point proceed vertically downwards and record the pressure drop shown there in bars.

The branch lines from the closed loop main line are normally of 25 mm size and the outlet connections at the points of consumption are generally of 19 mm size available for the final use of pneumatic tools.

14.2 *Valves for Compressed Air System* — Valves are installed in air distribution system so as to divide the system into a number of sections. Further, they are provided at the point of air consumption to isolate air-operated machines and tools.

Gate valves, diaphragm valves, globe valves, non-return valves, and plug valves, all find application in compressed air distribution network. Gate valves have very low pressure drop and

are used for isolation purposes. Globe valves (whether straight-way type of angle type) have high resistance and are used for controlling the flow. However, in small sizes, globe valves are also used for shut-off because of their better air-tightness. Other types of valves used for shut-off are diaphragm valves and plug cocks. The frictional resistance of diaphragm valves lies between gate valves and globe valves. Plug valves may not be fully leak-proof unless made of lubricated plug design, but the latter are costlier than gate valves and require more attention.

It is essential to provide a non-return or a check valve at the delivery end of a rotary compressor. In case of a number of reciprocating compressors connected to a single receiver this valve should be provided for delivery line of each compressor. Also, when no isolation valve is provided between a compressor and its receiver, a check valve should be installed. This helps to isolate the machine from the receiver for maintenance. Alternatively, fullway gate isolation valves may be provided but then it is most important to see that a safety valve of sufficient size is fitted on the compressor side of each isolation valve otherwise, severe damage may be caused if the valve is closed while the compressor is running. The friction losses in valves are given in Table 3.

15. Leakage Losses in Compressed Air Systems

15.1 Quantity of air losses through small holes, cracks, leaky couplings, joints, etc, can add up to a very large value. In old installations, where inspection and maintenance have been rather poor, a loss of 25 to 30 percent of the total compressor capacity is not unusual. With proper installation and maintenance leakage losses should not exceed 3 percent of the total capacity of the compressor. With the continued leakage joints, there is a rise in leakage percentage and affects the economic operation of the plant as a whole. Table 5 shows leakage losses in m^3/min, with the air pressure at 6 bars, for different sizes of holes together with the power consumption. As the air passing through a leak is proportional to the absolute pressure, the loss will be greater with higher pressure in the pipeline. Further, with bad leakages, the drop in pressure in the line may reach such limits as to effect the operation of pneumatic machines.

From the above it will be apparent, regular inspection and systematic maintenance are absolutely essential to completely eliminate leakage losses to reduce them to a minimum.

In shipyards, iron works, mines and quarries, the system is often large with many branch lines which are frequently temporary and in such cases a leakage loss up to even 10 percent may have to be tolerated.

Similarly, in forging shops using pneumatic hammers and in foundry using pneumatic moulding machines, it is difficult to keep the leakage below 10 percent as the control devices and working cylinders of these machines are subject to rapid wear and generally become leaky after relatively short period of use.

TABLE 5 RELATION BETWEEN HOLE SIZE, LEAKAGE AND POWER LOSS

Hole Diameter mm	Air Leakage at 6 Bars m^3/min	Power Required for Compression in kW
1	0.06	0.3
3	0.60	3.1
5	1.60	8.3
10	6.30	33.0

15.2 *Leakage Measurement*

a) Ensure that the piping is open all the way up to the control valves of all pneumatic equipment normally used, without making any changes in the system from its normal operating condition. All the control valves must be completely closed.

b) Measure the quantity of air delivered by the compressor plant at normal working pressure with no useful air consumption from the system. This gives the amount of leakage air for the system.

c) A compressor of known capacity is used as a measuring compressor and the other compressors in the plant are stopped. The measuring compressor has a capacity in excess of the expected total leakage. The measurement of leaked air is made as follows:

i) The compressor charges the system to approximately 7 bars when it should unload or be unloaded and a stop watch started.

ii) The leakage in the system will now cause a pressure drop to the lower operating limit say approximately 6.4 bars. When the lower operating limit is reached the compressor will start loading again. Record the time of start of loading without stopping the stop watch. Note the time also at the time of unloading the compressor again. Repeat this operation at least four times and record the time.

If the compressor capacity is Q m^3/min and the total load time in minutes, the delivered amount of air is $q \times t$ m^3. If the total measured running time is T minutes the average amount q of air leaked is:

$$q = \frac{Q \times t}{T} \text{ m}^3/\text{min}$$

The time measurement for the reloading period plays an important part in the assessment of leakage air. The start and finish of reloading is indicated by the pressure gauge reading and as such during this experiment a well calibrated pressure gauge should be used.

If a compressor with two-stage unloading is used in the test, it is sufficiently accurate for practical purposes to calculate with half the compressor capacity at half load.

In most of the shops where pneumatic machines are used, hose couplings, throttle valves and other fittings connected to the supply outlet frequently cause heavy leakage. In such cases it is preferable to carry out the above mentioned test with all the pneumatic machines connected as for normal operation and one additional test with the shut-off valves at the connections closed. The difference between these two measurements indicates the leakage of the pneumatic machines and their fittings. Thus this itself also helps to assess the condition of the various pneumatic machines used in the shops.

16. Operation and Maintenance

16.0 For efficient operation of a compressor, it is essential to have a periodic preventive maintenance. The frequency of checking, inspection and maintenance depends on a particular installation, the time for which the installation is run continuously, past service experience, etc. The maintenance of prime movers, which is not covered in this standard, should be carried according to manufacturer's recommendation.

16.1 Items which require constant attention are as follows:

a) Compressor supplying air at correct pressure, temperature and with normal power consumption;

b) Any unusual sound or vibration;

c) Oil pressure and temperature and oil level in the sump;

d) Proper oil feed to cylinders and packing (in case of force-feed lubricating systems);

e) Grease points properly checked;

f) Excessive temperature of bearings and cylinder valves;

g) Flow and temperature of cooling water in jackets and coolers;

h) Air leakages; and

j) Draining of condensate from coolers, receivers, traps, etc.

16.1.1 Other items which require frequent attention are as follows:

a) Intake filter,

b) Oil filter,

c) Change of oil, and

d) V-belt tension.

In addition, water jackets, fins, cooler tubes should be periodically cleaned. Safety devices, including setting of safety valves and foundation bolts to be checked periodically.

16.2 For the guidance of the user, a schedule for the maintenance of compressor installation is given below. However, it is not intended to constrain any of the recommendations given by the manufacturer in the maintenance manual.

16.2.1 *Reciprocating compressor*

a) Daily:

i) Check oil level in crankcase;

ii) Drain receivers, aftercoolers and intercoolers and any other cooling system;

iii) Safety valve to be checked by manual operation;

iv) Check the lubricating oil pressure;

v) Drain water from lubricating oil sump and top up with lubrication oil;

vi) Turn machine by hand with interstage drain open. At the same time give the cylinder lubricating oil pump a few strokes by hand; and

vii) Grease barrel of water pump.

b) After 50 h of running:

i) Bolts to be tightened and adjusted;

ii) Check suction filters;

iii) Check cooling water pump gland;

iv) Check stage pressure and temperature;

v) Check lubricating oil filter; and

vi) Change lubricating oil required for 50 h of running.

c) After 150 h of running:

i) Clean air inlet filters;

ii) Remove and clean lubricating oilstrainer in sump;

iii) Check engine lubricating oil system for leaks;

iv) Open engine crankcase cover and examine the screwing arrangement and slackness of big end, cross-head and main bearings, inspect screwing arrangement of all bearings, lubricating pipes and connections. Examine lubrication oil pumps;

v) Examine samples of lubricating oil from sump for metal particles and emulsification (if the examination indicates severe discoloration or emulsification it should be changed);

vi) Clean water pump suction weed trap; and

vii) Clean circulating water suction strainer (if fitted).

d) After 300 h of running:

i) Clean valve assembly;

ii) Check air governor for its functioning;

iii) Check safety valve setting for its correctness of pressure at which it should be blown;

iv) Clean lubricating oil sump and renew oil;

v) Examine engine big end, cross-head and main bearings and adjust oil clearances; and

vi) Check that the freedom of movement of compressor, if mounted on shock and vibration mountings, is not restricted.

e) After 1 000 h of running:

i) Check crankshaft alignment;

ii) Check crankshaft on moving parts, connecting rod bolts, nuts, split pins, etc;

iii) Examine for wear and adjust compressor big-end bearing, gudgeon pin and bush;

iv) Check aftercooler and intercooler for scaling if any;

v) On shock/vibration mountings, inspect condition of necol lacquer. Clean and reseat (if it is to be used);

vi) Remove and clean auto-clean strainer catridges;

vii) Clean out and inspect intercoolers and water spaces on cylinder, cylinder head, etc;

viii) Hydraulic test of all cylinders, intercoolers, oil coolers and water spaces;

ix) Examine samples of lubricating oil from sump for metal particles and emulsification (if found necessary, lubricating oil should be changed);

x) Open crankcase cover and examine the screwing arrangement and slackness of big end, cross-head and main bearings, connecting red bolts, balance weight bolts. Examine lubricating oil pump and lubricating pipe connections;

xi) Examine for all stages, suction, delivery valves for wear and damage. It is important that valves and seatings should have clean and unpitted surfaces. Valve springs to be inspected for damage and tension and replaced, if necessary;

xii) Examine big end bearing bolt for elongation; and

xiii) Examine and gauge for wear compressor plunger and cylinder.

f) After 2 000 h of running:

i) Check all moving parts for wear;

ii) Examine cylinder lubricating oil pump for wear, bell valves to be examined for pitting or corrosion and renewed. Check all valve springs for condition and tension;

iii) Open the main big end and cross-head bearings and examine, adjust bearings oil clearances;

iv) Examine connecting rod for crank and straightness;

v) Pressure gauge for air and oil to be recaliberated; and

vi) Clean the sump and renew oil.

g) After 5 000 h of running:

i) Dismantle and clean all parts and replace worn out parts as per recommendations from manufacturers. An approved cleaning chemical is to be used for the parts and care is also to be taken to dry completely the parts before assembling. If compressor shall be kept idle for longer periods, an anti-rust oil shall be used along with lubricating oil and compressor rotated 5 to 6 times after every 2 to 3 days to lubricated all sliding or moving parts; and

ii) Replace connecting rod bolts and nuts.

16.2.2 *Rotary Compressors*

a) Daily:

i) Check levels of fuel, oil in sumps and filters;

ii) Drain the moisture separator;

iii) Drain any other cooling system provided;

iv) Drain water from lubricating oil sump check level of oil in sump;

v) Give one turn to grease cups fitted to water pump and drive; and

vi) Turn compressor by hand or motor.

b) After 50 h of running:

i) Run compressor up to working pressure. Check performance;

ii) Clean and oil all joints;

iii) Drain and clean compressor sump. Clean lubricating oil strainer. Refill sump; and

iv) Check for oil leakage and the security of all keeps and fastenings.

c) After every 200 h of running:

i) Check suction filter elements and clean;

ii) Drain the receiver for any water collected at the bottom;

iii) Examine suction and delivery valves for wear or damage and refit as necessary;

iv) Check lifting of air reservoir relief valve at designed pressure;

v) Clean intercooler fins;

vi) Inspect condition of necol lacquer on shock/vibration mountings, clean and recoat if necessary (in shock/vibration mountings are not fitted, check fastenings and holding-down bolts for tightness);

vii) Examine and refit the air compressor;

viii) Strip, examine the bearings, adjust the clearances of gears, repack glands of circulating water pump; and

ix) Open and refit the circulating water pump discharge valve.

d) After every 500 h of running:

i) Drain compressor oil and refill with fresh oil recommended by the manufacturer;

ii) Change filter elements in oil filter; and

iii) Drain and clean compressor sump, clean lubricating oil strainer and refill sump.

e) After every 1 000 h of running:

i) Clean and check all hoses provided on compressor;

ii) Attend to moisture separator;

iii) Check separator elements and replace, if necessary;

iv) Water discharge to cooler, check control valve; and

v) Examine water side and clean tubes of water cooler.

f) After every 5 000 h of running:

i) Dismantle and clean the compressor;

ii) Non - metric blades be replated and mettalic plates be replaced/dressed, if worn out; and

iii) Replace the damaged parts and also the parts recommended by manufacturers.

16.3 Other items, which require special attention while the machine is being stripped completely for cleaning, adjustment of clearances, replacement of worn out parts, etc, are also listed as a general guide.

16.3.1 *Reciprocating compressor*

a) *Cylinders* — These should be checked for undue wear. Excessive wear or ovality may require rematching of cylinder and use of oversize rings or changing of cylinder liner.

b) *Pistons* — While reassembling, positive locking of piston to piston rod is very important and also the piston clearances.

c) *Piston rings* — The piston rings should not stick in their grooves. If a end gap is higher than recommended, these should be replaced.

d) *Valves* — Carbon deposits should be removed. Valve plate/strips and valve seats should be examined. Valve springs should be checked for correct tension. The valve lift should be checked. Before assembly, valve plates and seats should be lapped. If necessary, seats should be reground to remove dents. Valve parts to be thoroughly cleanded when reassembling.

e) *Main bearings, big end bearings, gudgeon pin* — Check for wear, scoring, etc; scrap, if necessary. Adjust clearance as recommended.

f) *Oil Pump* — Check for worn out bushes and proper clearances between rotors/gears.

g) *Oil cooler, intercooler and aftercooler* — Test hydraulically against leakage at tubes once in 2 to 3 years, or earlier, if leakage is suspected.

h) *V-belts* — To be changed in further tensioning not possible. Change the whole set (*see* **9.4.4**).

16.3.2 *Rotary compressors* — It is important to provide correct clearance between rotors as per manufacturer's recommendation by adjusting the timing gears. Mechanical shaft seals should be checked for their sealing surfaces.

16.3.3 *Rotor blades of sliding vane compressors* — Check for blade wear and dress blades, if necessary.

17. Conversions

17.1 *Conversion of Actual Cubic Meter Per Hour (FAD) Wet to Kilogram Per Hour (Dry)*:

$$W(dry) = Pa \times V(wet) \times \frac{(P_a - P_p \times RH)}{P_a} \times \frac{MW}{T_1} \times 12.31$$

where

$W(dry)$	= mass of dry air	– kg/h
P_a	= suction pressure of compressor	– bar
$V(wet)$	= FAD (wet) of compressor	– m^3/h
P_p	= partial water vapour pressure at ambient temperature	– bar
RH	= relative humidity	– Fraction
MW	= molecular weight of wet air	– Dimensionless
T_1	= ambient temperature	– °K

17.2 *Conversion of Actual M^3/min (FAD) (Wet) to NM^3/min (Dry)*

$$V_2(dry) = V_1(wet) \times \frac{(P_a - P_p \times RH)}{(P_a)} \times \frac{P_a}{P_2} \times \frac{T_2}{T_1}$$

where

$V_2(dry)$	= volume of air at normal conditions	– NM^3/hr
	normal conditions	– N
	pressure	– 1.012 7 bar
	temperature	– 273.15 °K
	RH	– 0 Percent
$V_1(wet)$	= FAD (wet) of compressor	– m^3/min
P_a	= suction pressure of compressor	– 6 bar
P_2	= pressure at normal conditions bar, that is, 1.012 7 bar	– bar
P_p	= partial water vapour pressure at ambient temperature	– bar
T_1	= ambient conditions	– °K
T_2	= Temperature at normal conditions that is, 273.15 °K	– °K
RH	= relative humidity	– Fraction

17.3 *Conversion from Actual M^3/h (FAD) Wet to SM^3/h (Dry)*

Here symbols have the same meaning as in **17.2 except:**

$T_2 = 273.15 + 20 = 293.15$ °K

Standard conditions 'S'

Pressure	— 1.012 7 bar
Temperature	— 293.15 °K
RH	— 0 Percent

$$V_2\ (\ dry\ ) = V_1\ (\ wet\ ) \times \frac{P_a - P_p \times RH}{P_a} \times \frac{P_a}{1.012\ 7} \times \frac{293.15}{T_1}$$

APPENDIX A

(*Clauses* 4.3 *and* 4.4)

EXAMPLE OF CALCULATION

The following calculation is typical of a medium-sized engineering workshop including a foundry, where a high degree of mechanization is to be carried out by means of compressed air-driven machines and tools. Listed in the table are the tools and other pneumatic devices which are expected to be included in the installation at full production capacity. The use factors for the different tools have been calculated in consultation with production planning and it has thus been possible to establish the average total air consumption.

Machine or Tool	Quantity	Air Consumption Per Unit m^3/min	Maximum Air Consumption m^3/min	Use Factor	Average Air Consumption m^3/min
Foundry					
I) Core-shop:					
Core blowers	3	0.65	1.95	0.50	0.98
Bench rammers	2	0.30	0.60	0.20	0.12
			2.55		1.10
II) Machine moulding:					
Moulding machines	5	0.70	3.50	0.30	1.05
Blow guns	5	0.50	2.50	0.10	0.25
Air hoist — ½ ton	2	2.00	4.00	0.10	0.40
			10.00		1.70
III) Hand moulding:					
(Rammers)					
Ramdium	1	0.35	0.35	0.20	0.07
Heavy	2	0.55	1.10	0.20	0.22
Blow guns	3	0.50	1.50	0.10	0.15
Air hoist — ½ ton	1	2.00	2.00	0.10	0.20
			4.95		≈ 0.65
IV) Cleaning shop:					
a) Chipping hammers					
light	2	0.35	0.70	0.35	0.25
medium	3	0.50	1.50	0.35	0.52
heavy	2	0.75	1.50	0.20	0.30

Machine or Tool	Quantity	Air Consumption Per Unit m^3/min	Maximum Air Consumption m^3/min	Use Factor	Average Air Consumption m^3/min
b) Grinders:					
75 mm	2	0.55	1.10	0.30	0.30
150 mm	3	1.50	4.50	0.45	2.03
200 mm	1	2.40	2.40	0.20	0.48
medium	2	1.40	2.80	0.10	0.28
heavy	2	2.50	5.00	0.10	0.50
c) Sand blast units:					
light	1	1.90	1.90	0.50	0.95
heavy	1	3.20	3.20	0.50	1.60
			24.60		≈ 7.20
Total for foundry			42.10		10.65
Workshop					
V) Machine shop:					
Blow guns	10	0.50	5.00	0.05	0.25
Operating cylinders for jigs, fixtures and chucks			0.75	0.10	0.08
			5.75		≈ 0.35
VI) Sheet metal shop:					
a) Drills					
light	1	0.35	0.35	0.20	0.07
medium	1	0.45	0.45	0.20	0.09
13 mm	2	0.90	1.80	0.30	0.54
angle	1	0.45	0.45	0.20	0.09
screw-feed	1	3.10	3.10	0.05	0.15
b) Taper	1	0.45	0.45	0.20	0.09
c) Screw drivers	2	0.45	0.90	0.10	0.09
d) Impact wrench:					
20 mm	1	0.90	0.90	0.20	0.18
22 mm	1	1.35	1.35	0.10	0.14
e) Grinders:					
150 mm	2	1.50	3.00	0.30	0.90
200 mm	1	2.40	2.40	0.20	0.48
medium	2	1.40	2.80	0.30	0.84
heavy	1	2.50	2.50	0.20	0.50
f) Riveting hammers:					
medium	1	1.10	1.10	0.10	0.11
heavy	1	1.30	1.30	0.05	0.07

Machine or Tool	Quantity	Air Consumption Per Unit m^3/min	Maximum Air Consumption m^3/min	Use Factor	Average Air Consumption m^3/min
g) Chipping hammers:					
light	2	0.35	0.70	0.20	0.74
medium	2	0.50	1.00	0.20	0.20
heavy	1	0.75	0.75	0.10	0.07
air hoist — 5 ton	1	5.80	5.80	0.05	0.29
blow guns	2	0.50	1.00	0.10	0.10
			32.10		5.75
VII) Assembly shop					
a) Drills:					
light	3	0.35	1.05	0.20	0.21
medium	5	0.45	2.25	0.30	0.68
13 mm	6	0.90	5.40	0.35	1.89
angle	2	0.45	0.90	0.10	0.09
heavy	1	1.30	1.30	0.10	0.13
heavy	1	2.00	2.00	0.10	0.20
b) Tapers	2	0.45	0.90	0.10	0.09
c) Screw drivers	2	0.45	0.90	0.20	0.18
d) Impact wrenches:					
light	1	0.35	0.35	0.20	0.07
20 mm	2	0.90	1.80	0.20	0.36
22 mm	1	1.35	1.35	0.10	0.14
e) Grinders:	2	0.41	0.90	0.10	0.09
75 mm	2	0.55	1.10	0.20	0.21
150 mm	1	1.50	1.50	0.10	0.15
medium	2	1.40	2.80	0.20	0.56
f) Air hoists:					
½ ton	1	2.00	2.00	0.10	0.20
1 ton	1	2.00	2.00	0.10	0.20
g) Blow guns	5	0.50	2.50	0.05	0.13
			31.90		≈ 5.60
VIII) Painting shop					
Grinders and polishers					
angle	1	0.45	0.45	0.20	0.09
medium	1	1.40	1.40	0.30	0.42
Sand-blast unit	1	2.30	2.30	0.50	1.15
Blow guns	1	0.50	0.50	0.10	0.05
Air hoist — 5 ton	1	5.80	5.80	0.05	0.29
Spray painting guns	2	0.30	0.60	0.50	0.30
			11.55		2.30
Total for the workshop			81.30		14.00

Foundry

I) Core shop	1.10 m^3/min
II) Machine moulding	1.70 m^3/min
III) Hand moulding	0.65 m^3/min
IV) Cleaning shop	7.20 m^3/min
	10.65 m^3/min

Machine or Tool	Quantity	Air Consumption Per Unit m³/min	Maximum Air Consumption m³/min	Use Factor	Average Air Consumption m³/min
Workshop					
V) Machine shop	0.35 m^3/min				
VI) Sheet metal shop	5.75 m^3/min				
VII) Assembly shop	5.60 m^3/min				
VIII) Painting shop	2.30 m^3/min				
	14.00 m^3/min				

Note 1 — All tools are assumed to be operating between 6-7 bars.

Note 2 — Use factor indicates the estimated time of operation, that is, degree of utilization. This varies with condition of test, size of the workshop and number of tools in that category.

EXPLANATORY NOTE

First time this Indian Standard was issued in 1971 with a view to help the purchaser to device the capacity he needs, to meet a given set of requirements, install the selected compressor in the right place and in the right manner, and maintain it in such a way as to keep the cost per cubic metre of compressed air at the required operating pressure, to minimum. The intention of this standard is only to give general factors, which should necessarily be considered while selecting a suitable type and make. The responsibility of selecting a suitable type and make rests with the purchaser.

The purpose of revision of this standard is to update it keeping in view the present practice in the industry along with the experience gained in implementation of this standard during last twelve years.

This edition 2.1 incorporates Amendment No. 1 (September 1990). Side bar indicates modification of the text as the result of incorporation of the amendment.

AMENDMENT NO. 2 MAY 2003
TO
IS 6206 : 1985 GUIDE FOR SELECTION, INSTALLATION AND MAINTENANCE OF AIR COMPRESSOR PLANTS WITH OPERATING PRESSURES UP TO 10 BARS

(First Revision)

[*Page* 21, *clause* 14.1, *formula under informal table* (*see also Amendment No.* 1)] — Substitute the following for the existing formula:

The pressure drop in the pipeline can be calculated by the following formula:

$$p = \frac{7.57 \times Q1.85 \times L \times 10^{-10}}{d^5 \times P}$$

where

p = pressure drop in the pipeline, bar;
q = air flow quantity, m^3/min (FAD);
L = lengths of the pipeline, m;
d = inside diameters of the pipeline, in mm, conforming to IS 1239; and
P = initial absolute air pressure, bar.

(ME 22)

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